

Structural Considerations of a 20MW Multi-Rotor Wind Energy System

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Abstract. The drive to upscale offshore wind turbines relates especially to possible reductions in O&M and electrical interconnection costs per MW of installed capacity. Even with best current technologies, designs with rated capacity above about 3 MW are less cost effective ex-factory per rated MW (turbine system costs) than smaller machines. Very large offshore wind turbines are therefore justified primarily by overall offshore project economics. Furthermore, continuing progress in materials and structures has been essential to avoid severe penalties in the power/mass ratio of large multi-MW machines. The multi-rotor concept employs many small rotors to maximise energy capture area with minimum system volume. Previous work has indicated that this can enable a very large reduction in the total weight and cost of rotors and drive trains compared to an equivalent large single rotor system. Thus the multi rotor concept may enable rated capacities of 20 MW or more at a single maintenance site.

Establishing the cost benefit of a multi rotor system requires examination of solutions for the support structure and yawing, ensuring aerodynamic losses from rotor interaction are not significant and that overall logistics, with much increased part count (more reliable components) and less consequence of single failures are favourable. This paper addresses the viability of a support structure in respect of structural concept and likely weight as one necessary step in exploring the potential of the multi rotor concept.

1. Introduction

The primary benefits of a multi-rotor system relate to:

- 1) *Scaling laws* – the total sum of rotors and drive trains of the multi rotor system can have much less weight and cost compared to a single equivalent turbine.
- 2) *Standardisation* – systems larger than 20 MW will be realised with more rotors and not larger rotors. Standardising rotor and drive train components will allow for stable serial production at a size comfortably within industry experience. This in turn will lead to very substantial cost reductions and improvements in reliability.
- 3) *Maintenance* – the multi-rotor system will have in effect almost no unscheduled maintenance. Single turbine faults will usually compromise only a few percent of capacity, reducing urgency to find favourable weather windows for remedial action.

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Industry effort is presently focussed on very large-scale single wind turbines, with ratings in the 5-10MW range, particularly for use in offshore locations. Recent studies (UPWIND[1]) have proposed single rotor machines as large as 20MW in rating, following obvious cost savings associated with minimizing the number of offshore foundations required for a given total power output. A multi-rotor wind energy system could offer similar power outputs (or even greater) without escalating structural penalties and at a reduced cost of energy (CoE). Analysis based on scaling with similarity [2] predicts that the total weight of rotor mass and major drive train components scale as $\frac{1}{\sqrt{N}}$ where N is the number of rotors in a multi rotor system. Such scaling implies cubic variation of aerodynamic bending moments and mass of many major components, thus many small rotors may have much less mass and cost than an equivalently rated single rotor. While it is an ongoing challenge to improve technology to avoid cubic up-scaling, it merely requires the application of state of the art capability to downscale cubically.

The multi-rotor concept is not new, appearing in designs as early as the 1930s [2]. However, the initial concept was a response to the difficulty of making very large rotors when only steel was considered a reliable structural material, a problem mitigated now by advances in composite materials. The multi-rotor concept offers a route to much larger unit capacities than is currently possible with single rotors and without the adverse impact of upscaling on turbine CAPEX and increasing CoE.

A preliminary 20MW multi-rotor system has been conceived which places 45, 444kW rotors on a single structure utilising currently available technology. Initial work by Jamieson [3] suggests that the cost advantage of such a multi-rotor system can yield CAPEX savings of 11% compared to 4x5MW rotors and 30% compared to a single 20MW machine - assuming that material costs are closely tied to mass.

To validate this theory further, this paper presents one potential structural layout which would meet the energy capture and structural capacity required for a 20MW multi-rotor system in an off-shore environment. From this, more useful comparisons can be made to other alternative options and help solidify the argument for the further research and development of the multi-rotor concept.

2. Structural Considerations

Multi-rotor systems (MRS) do not require any technology or engineering practices that are not currently in use today. However, the full economic potential of the multi-rotor system will only be realised by developing optimised designs of wind turbine systems that benefit from the technological progress that is presently being applied to the largest turbines.

The geometric scale of the system will necessarily be similar to that of an equivalently rated single rotor system given that modern designs approach theoretical power capture limits. However, the advantage of using small, well-designed rotors might yield 78% savings on rotor and drive trains mass compared to a single rotor obeying the square/cubed power law and 58% compared to four 5MW machines if cost is closely tied to mass [3].

Previous confidential studies have shown that the critical extreme loads on a 2MW 7-Rotor MRS were generally less than for an equivalent single rotor system. Moreover the weight of a frame to support the rotors plus all the rotors and drive-trains may still be less than the weight of 3 blades of a single rotor. A conceptual solution therefore is to support the frame and multi-rotor system on essentially the same tower and yaw ring as the equivalent large single rotor. This solution was investigated but is considered sub-optimal and not pursued here. However it reveals clearly that providing for yawing of a multi-rotor system should not be a critical threat to its economic potential benefit.

At present there are no existing commercial examples of multi-rotor systems although proto-type sub-MW designs have been constructed in the past. Examples of dual/tandem rotors exist, but these systems look to take advantage of effects other than those considered here and therefore should not be classified under the same category.

methods employed, e.g. welding techniques, are transferrable. The structural design is affected by self-weight, extreme wind loading and for an offshore environment wave loading, though the latter will not be considered here. Wind loading can quickly become a design limiting factor when considering IEC Class I 50-year storm conditions and therefore necessitates a careful consideration of the type and thickness of members used to join adjacent nodes.

Given the manufacturing complexity and operational complexity of ensuring aerodynamic members are properly aligned – streamlined bodies as load bearing members will not be considered. This leaves the circular cross-section as the only realistic alternative (in some cases possibly with aerodynamic fairings). The trade-off is then between strength to mass ratios and drag of circular hollow sections (CHS). For the purpose of this study a numerical optimisation problem was carried out in Mathematica by varying the outer and inner diameter of these hollow sections in an attempt to minimize mass and maintain mechanical strength under increasing axial loading during an extreme storm case $V_{e50} = 50\text{m/s}$.

Figure 2 shows this optimisation problem over a range of axial compression loads from 0 to 100,000kN, which was calculated as the worst-case loading for a rotor under any IEC-61400 conditions. The results show a clear region above outer diameters of 1.8m in which increasing diameter causes negative effects due to increased drag. The results also show that member widths around 0.8m allow for relatively thin sections that ultimately offer the best strength to mass ratios and provide a basis for the initial structural layout.

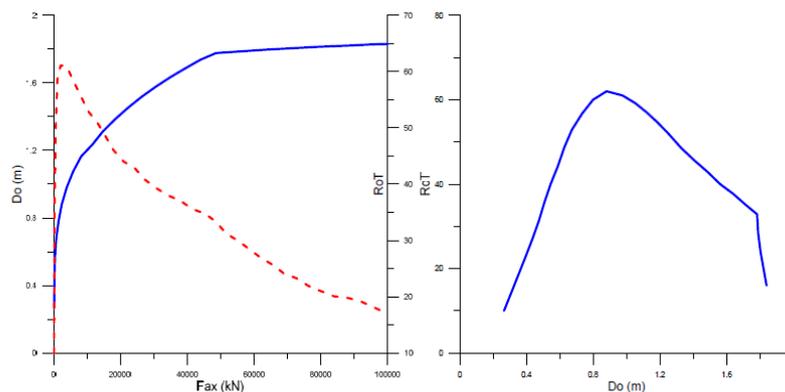


Figure 2. Left: Optimising Sectional Properties as a Function of Axial Compression Load (F_{ax}), Right: Plot of Outer Diameter vs. the Radius Over Thickness Ratio Simulation

3.2. Design Requirements

IEC-61400-1 (2005) [6] was used as the basis for providing design limiting load cases to test against the proposed MR system. All 15 ultimate load cases were input into the most recent version of GH Bladed which allows for the modelling and simulation of 45 rotors on a single structure. The hub loads F_x , F_y , F_z , M_x , M_y and M_z were analysed at each individual rotor as well at the structures base. Summing the forces from each rotor provides the tower top loading for comparison against an equivalent 20MW single rotor. The load case corresponding to the ultimate load for each variable was found and a set of reduced load cases; DLC1.3, DLC2.3 and DLC 6.1/2 produced. These load cases were then run in more depth.

Single rotor faults, such as one blade stuck in pitch which introduce major unbalanced loads on a single turbine, have little impact on the structure design of a MR system and therefore such fault load cases were discounted.

3.3. Ultimate Loading

Normal operation in extreme turbulence (IEC DLC1.3 [6]) is often a designing load case for major components of large wind turbines. This load is also the most significant of the rotor derived loads for the multi-rotor system structure. However, because the turbines all operate at slightly different speeds

and blade angles, this destroys much of the coherent loading on the support structure. The extreme base bending moment, M_y , is about 30 times less than for an equivalent single rotor and fatigue loads on the structure arising from rotor loading are substantially erased to a low level of white noise through destructive interference (Figure 3). To be clear each individual turbine is designed for the usual loads it would experience as a single unit turbine but the aggregate loading passed on to the structure is much less significant.

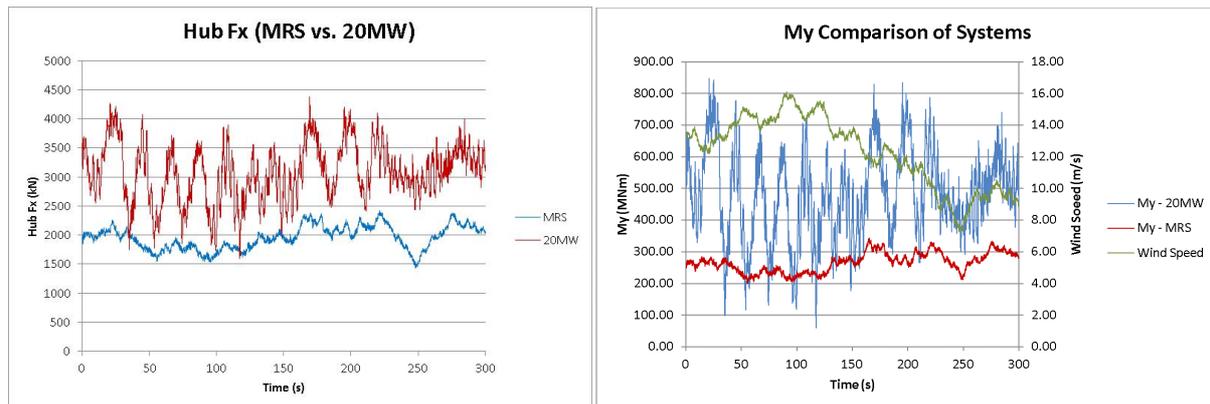


Figure 3. Left: Total Hub Fx (MRS vs Single Rotor), Right: Total Base My (MRS vs Single Rotor)

The only load case that caused concern for the multi-rotor system was grid loss. This is because the relatively coherent shutdown of all the turbines that would occur produces transient loading similar to that of a single 20 MW turbine, Figure 4. This is not fundamentally problematic but it would clearly be disappointing to lose all the major comparative advantages in loads and to have this as a designing load case for the support structure. By using dump resistors to vary delay in the total loss of load of each turbine and without much added cost or weight, the coherence in shut down is obviated and grid loss was no longer a designing load case. In consequence a remarkable situation is achieved. At 20 MW capacity or greater an effective wind turbine system can be produced in which rotor loading is never designing for the support structure. The support structure is designed by aerodynamic forces from storm wind loading on its own members.

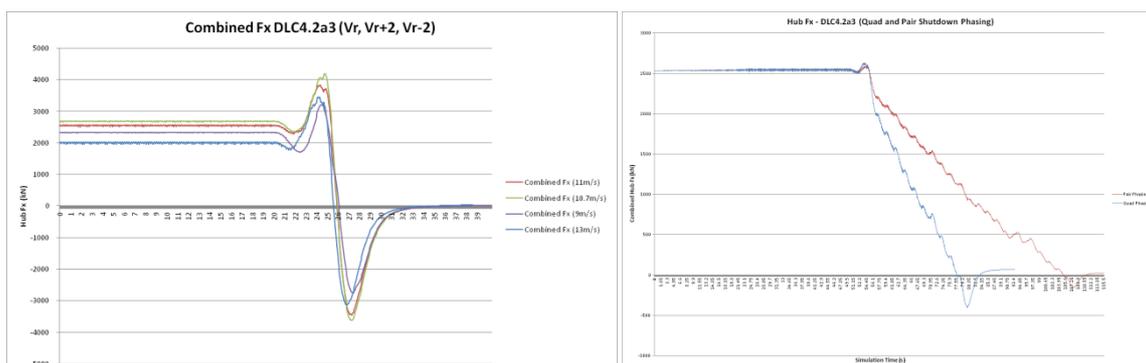


Figure 4. Left: Fault Case Shutdown Fx Loading, Right: Staged Shutdown Fx Loading

4. Structural Optimisation

The space frame was constructed in the finite element analysis package Abaqus 6.8-3. The depth of the frame is optimised for minimum mass and tapered from 3.577m (the depth of the nacelle) to 30m at the base of the space frame. Rotors are modelled as a distributed nacelle load of 8t and a F_y point load representing a 3 tonne rotor, bolted into a square bracket of solid steel. A time-varying pressure load is applied to the front face of each nacelle to represent dynamic thrust loading as taken from corresponding Bladed simulations. Time varying moments are applied to each nacelle representing a

rotor idling at 0.05 rad/s (for the extreme storm cases), with adjacent rotors rotating in opposite directions. Line loads corresponding to a constant 50m/s extreme wind speed are placed on all structural members. A coefficient of drag for the CHS is taken to be 1.2, though this may be lower in reality. The extreme wind case is carried out in 10 degree increments around the entire structure and a safety factor of 1.35 applied to all the loads.

The structure was set up as per the initial concept and using the thickest type of member calculated in Mathematica and using the worst-case Fx loading as taken from Bladed and then iterated with succesively decreasing member size until one or more structural failure. In this way, the structural members are optimised for minimum mass and structural integrity in the worst-case loads.

To simplify the engineering of a large space frame, a small number of circular cross-sections are selected from all those determined during optimisation (Table 1). The near optimum layout of these CHS are presented in Figure 5, with each set labelled.

Table 1. List of structural members and properties. S is the section modulus and M_{yield} the yield strength under bending.

Beam ID	Outer Diameter (m)	Thickness (m)	Mass per Member (tonnes)	Number of Members	S (I_y/y)	M_{yield} (MN/m)
CHS3	0.6096	0.0064	4.04	8	0.0018	0.642
CHS4	0.7112	0.0064	4.73	24	0.0025	0.878
CHS5	0.8128	0.0071	5.99	26	0.0036	1.273
CHS6	0.9144	0.0079	7.50	32	0.0051	1.794
CHS7	1.016	0.0087	9.18	30	0.0069	2.439
CHS8	1.117	0.0095	11.02	20	0.0091	3.220
CHS9	1.219	0.0111	14.05	36	0.0126	4.472
CHS10	1.320	0.0127	17.39	12	0.0169	5.991
CHS11	1.422	0.0191	28.07	14	0.0291	10.337
CHS12	1.524	0.0254	39.88	8	0.0440	15.636
CHS13	1.625	0.0318	53.07	10	0.0621	22.063

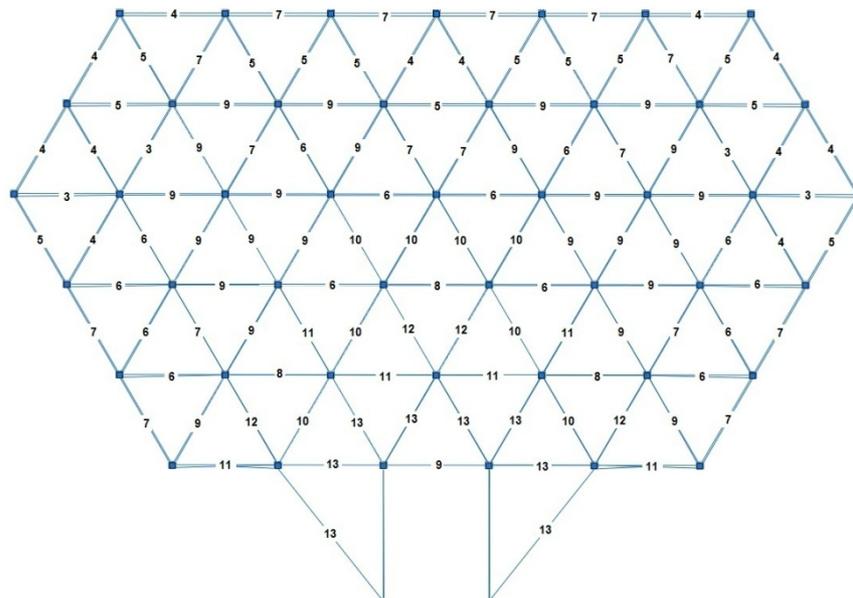


Figure 5. Front Elevation of MRS with CHS Members Labelled

4.1. Structural Analysis

An Eigenvalue analysis of the multi-rotor space frame was carried out by CRES, Athens and the first 200 of these Eigenvalues are presented in Table 2. This analysis forms the basis of future work examining fatigue loading on the structure and is presented only briefly here.

The proposed space frame has a wide range of closely spaced natural frequencies, some of which lie within the main excitation frequency range of the rotors (0.44 Hz to 0.78 Hz). However the rotor frequency varies randomly and destructive interference of the aggregate input from the rotors to the structure leads to vastly reduced fatigue load ranges compared to a single large rotor (see again Figure 3). Thus it seems unlikely that undue resonant response and fatigue loading in general will be problematic. Final fatigue load calculations are presently being conducted by CRES.

Table 2. First 200 Eigenvalues of the proposed multi-rotor space frame with Eigenvalues in the 1P operational range highlighted in bold.

N/N	Hz																		
1	0.35	21	1.33	41	2.11	61	3.01	81	3.55	101	4.18	121	4.81	141	5.78	161	6.69	181	7.75
2	0.39	22	1.35	42	2.16	62	3.02	82	3.60	102	4.23	122	4.82	142	5.86	162	6.75	182	7.90
3	0.49	23	1.47	43	2.24	63	3.08	83	3.61	103	4.24	123	4.85	143	5.88	163	6.78	183	7.92
4	0.56	24	1.50	44	2.27	64	3.09	84	3.65	104	4.28	124	4.91	144	5.92	164	6.82	184	7.92
5	0.59	25	1.51	45	2.30	65	3.15	85	3.66	105	4.30	125	4.97	145	5.94	165	6.85	185	7.97
6	0.66	26	1.54	46	2.38	66	3.19	86	3.75	106	4.33	126	5.04	146	6.04	166	6.90	186	8.16
7	0.67	27	1.68	47	2.45	67	3.20	87	3.76	107	4.35	127	5.06	147	6.06	167	7.01	187	8.23
8	0.75	28	1.70	48	2.46	68	3.25	88	3.80	108	4.38	128	5.08	148	6.12	168	7.04	188	8.33
9	0.78	29	1.71	49	2.52	69	3.26	89	3.84	109	4.40	129	5.12	149	6.15	169	7.10	189	8.35
10	0.80	30	1.72	50	2.57	70	3.27	90	3.87	110	4.43	130	5.14	150	6.24	170	7.12	190	8.37
11	0.83	31	1.77	51	2.62	71	3.28	91	3.92	111	4.50	131	5.23	151	6.26	171	7.14	191	8.43
12	0.85	32	1.80	52	2.65	72	3.30	92	3.94	112	4.54	132	5.30	152	6.32	172	7.18	192	8.48
13	0.89	33	1.83	53	2.69	73	3.32	93	3.95	113	4.55	133	5.30	153	6.36	173	7.36	193	8.52
14	0.90	34	1.89	54	2.74	74	3.34	94	3.95	114	4.56	134	5.40	154	6.39	174	7.44	194	8.59
15	0.93	35	1.96	55	2.75	75	3.36	95	3.99	115	4.59	135	5.42	155	6.46	175	7.45	195	8.61
16	0.99	36	1.97	56	2.78	76	3.38	96	4.08	116	4.60	136	5.55	156	6.48	176	7.47	196	8.69
17	1.01	37	2.01	57	2.81	77	3.38	97	4.11	117	4.64	137	5.58	157	6.55	177	7.51	197	8.81
18	1.13	38	2.05	58	2.86	78	3.43	98	4.15	118	4.66	138	5.63	158	6.57	178	7.55	198	8.91
19	1.15	39	2.07	59	2.90	79	3.47	99	4.15	119	4.76	139	5.65	159	6.64	179	7.68	199	9.02
20	1.30	40	2.09	60	3.00	80	3.50	100	4.16	120	4.80	140	5.73	160	6.68	180	7.70	200	9.05

4.2. Results

The total space frame mass as calculated approximates 3000 tonnes excluding the tower. The addition of 45 rotor nacelles (11t per rotor) and expected mass increases due to joint welds, brackets etc (1t per rotor) brings the total tower head mass to approx. 3540 tonnes. This is equivalent to a cubically up-scaled (with similarity) ø274m single rotor and drive-train. In comparison, the UPWIND design (ø252m rotor) predicts that a tower head mass of 880t and total system mass of 3640t including tower is achievable. To achieve a fair comparison in terms of power density, the UPWIND design would nominally require a 274m rotor. Following scaling laws, this would result in a $(274/252)^3 = 1.3$ increase in tower mass. Thus the MRS system should perhaps be compared to the UPWIND mass estimates inflated by 30%.

A comparison of this data, including the original 5MW mass data is presented in Table 3. Note that in this example the multi-rotor tower is simply the members connecting the space frame to the ground and not a tubular tower, hence the reduced mass.

Considering the substantial savings in total cost of rotors and drive trains due to downscaling and the reasons cited in section 1, the CoE of the multi rotor system is unlikely to be penalised by excessive mass or cost in the multi rotor structure.

Table 3. Mass comparison of three system types. The 5MW reference is based on current 5MW machines and is multiplied by 4 to achieve 20MW. The 20MW single rotor is scaled up from the 5MW reference with similarity, each 444kW rotor on the MRS is scaled down from the 5MW reference with similarity. The 20MW UPWIND is an advanced conceptual design, scaling down to 444kW with similarity achieves an equivalent MRS system.

	5MW Reference x 4	20MW Scaled	20MW MRS Scaled	20MW UPWIND Design [1]	20MW MRS based on UPWIND
Tower Head Mass (t)	1392	2300	3500	880	2850
Tower Mass (t)	2210	3500	400-600	2760	400-600
Total System Mass (t)	3602	5700	3900-4100	3640	3250-3450

5. Discussion

The distributed nature of loading from multi-rotors at a scale of 20 MW and greater (assuming larger systems have more rotors and not larger rotors) appears to lead to a structure which for Class 1 offshore sites will be designed by extreme wind loads on its own members and not by loading input from the rotors.

Fatigue load calculations for the MRS structure have yet to be completed but the very low load ranges resulting from total rotor loading strongly suggest that extreme storm loads rather than fatigue loading will drive structure design.

Initial results suggest that the current MRS concept can feasibly achieve the aim of 20MW rated power without being adversely limited by design critical loads, particularly extreme turbulent storm cases which often design limit large single rotors. The current space-frame and power-train mass is equivalent to a notional 20MW machine up-scaled from 5MW with similarity (Table 3).

This suggests that the large COE benefit of a MRS system associated with reduced rotor and drive train cost will not be significantly compromised by adverse structure cost.

Furthermore, the MRS system can take advantage of several factors that a single rotor cannot. For example, increased degrees-of-freedom, quicker response to varying wind fields and importantly increased use of standardised components which help drive down the cost-of-energy.

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